3D Flow Simulation of a Spiral-Grooved Turbo-Molecular Pump

Saburo Igarashi*

*Iwaki Meisei University, Iino Chuohdai, Iwaki City, 970-8551, JAPAN

Abstract. A spiral-grooved turbo-molecular pump is widely used in vacuum technology. Available rotational speed of a rotor becomes larger and larger, and reaches 18,000 to 50,000 rpm. The clearance effect between a rotor and its casing is one of the important problems for the performance prediction of turbo-molecular pump which operates under very high speed of revolution. Now the pumping performance can be predicted from the computation of the three-dimensional flows. In the present paper, a flow in a spiral groove of turbo-molecular pump is simulated on the basis of the NS equations to cover a wide range of pressure ratio and mass flow rate. A fully three-dimensional analysis is made to take account of the effect of a gap between a rotor and its casing as well as the centrifugal and Coriolis forces. The pumping performance is shown in comparison with that of the no-gap flow and the channel flow approximations.

INTRODUCTION

A spiral-grooved turbo-molecular pump as shown in Fig.1 is widely used to cover a wide range of pressure ratio and flow rate in vacuum technology. Available rotational speed of a rotor reaches 18,000 to 50,000 rpm. The performance of a pump can be now predicted by computational fluid dynamics. The high rotational speed gives us many problems such as a large clearance between a rotor and its casing. But the flow is in low density and then may be considered to be laminar. In a very low pressure such as 1 Pa or less, the flow must be simulated by the DSMC method [1]. In the past, a flow in a groove was simulated by the 3D channel flow approximation (see Fig.2) by Nanbu et al. [2] in continuum model. The approximation cannot not exactly predict the performance of a pump for such a high rotational speed, as was suggested by the BGK model calculation of Kanki [3]. The effect of the centrifugal and Coriolis forces on the performance a of turbo-molecular pump is simulated by the NS equations (Igarashi [4]), where the effect of gap δ in Fig.3 between a rotor and its casing is neglected.

In the present paper, the effect of the clearance on the performance of a pump, that is, pressure and discharge rate distributions averaged over the cross section and on flow patterns is clarified by the NS equations with constant temperature approximation.

BASIC EQUATIONS AND CALCULATION METHOD

We consider the motion in the coordinate system fixed on the rotor whose angular velocity is $\boldsymbol{\omega}$ with the components $(\omega, 0, 0)$. The absolute velocity $\boldsymbol{u}$ is related with the relative one $\boldsymbol{w}$ by the formula $\boldsymbol{u} = \boldsymbol{w} + \boldsymbol{\omega} \times \boldsymbol{r}$. The relative motion is described on the assumption that the temperature change is negligible and then the pressure p is given by the equation $p = R\rho T$(R is the gas constant per unit mass), as was done in the previous paper [4]. The governing equations are in the cylindrical coordinate system as follows [5,6]:

$$\frac{\partial U}{\partial t} + L(U) = 0, \quad \text{where} \quad L(U) = \frac{\partial E_1}{\partial x} + \frac{\partial E_2}{\partial r} + \frac{1}{r}\frac{\partial E_3}{\partial \theta} + \frac{H}{r} - \frac{\partial S_1}{\partial x} - \frac{\partial S_2}{\partial r} - \frac{1}{r}\frac{\partial S_3}{\partial \theta} - \frac{S_4}{r},$$

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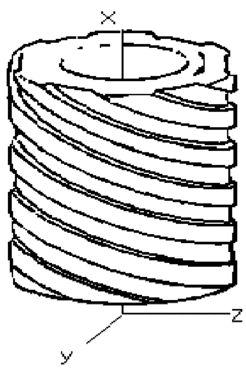

FIGURE 1. Spiral groove

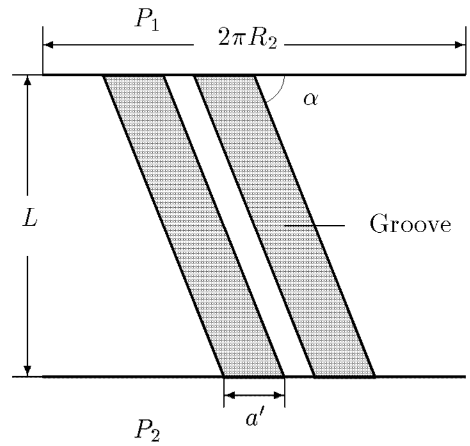

FIGURE 2. Plane view of rotor at the outer radius $r = R_2$

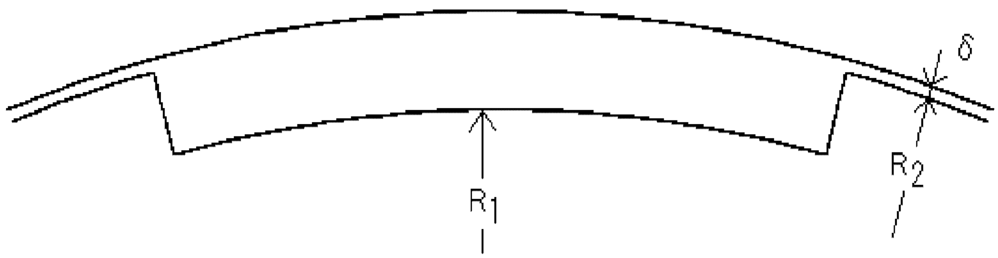

FIGURE 3. Calculation region of the cross section x=constant

$$U = \begin{pmatrix} \rho \\ \rho w_x \\ \rho w_r \\ \rho w_\theta \end{pmatrix}, \quad E_1 = \begin{pmatrix} \rho w_x \\ \rho w_x^2 + p \\ \rho w_x w_r \\ \rho w_x w_\theta \end{pmatrix}, \quad E_2 = \begin{pmatrix} \rho w_r \\ \rho w_r w_x \\ \rho w_r^2 + p \\ \rho w_r w_\theta \end{pmatrix}, \quad E_3 = \begin{pmatrix} \rho w_\theta \\ \rho w_\theta w_x \\ \rho w_\theta w_r \\ \rho w_\theta^2 + p \end{pmatrix},$$

$$S_1 = \begin{pmatrix} 0 \\ \Pi_{xx} \\ \Pi_{rx} \\ \Pi_{\theta x} \end{pmatrix}, \quad S_2 = \begin{pmatrix} 0 \\ \Pi_{rx} \\ \Pi_{rr} \\ \Pi_{r\theta} \end{pmatrix}, \quad S_3 = \begin{pmatrix} 0 \\ \Pi_{\theta x} \\ \Pi_{\theta r} \\ \Pi_{\theta\theta} \end{pmatrix}, \quad S_4 = \begin{pmatrix} 0 \\ \Pi_{rx} \\ \Pi_{rr} - \Pi_{\theta\theta} \\ 2\Pi_{r\theta} \end{pmatrix},$$

$$H = \begin{pmatrix} \rho w_r \\ \rho w_r w_x \\ \rho[(w_r^2 - w_\theta^2) - \omega^2 r^2 - 2r\omega w_\theta] \\ 2\rho(w_r w_\theta + r\omega w_r) \end{pmatrix}.$$

It is to be noted that U consists of density and three mass flux components, but has no temperature or energy term. Π_{ij} in the viscous terms of $S_i (i = 1-4)$ is the stress tensor, but it does not include pressure term, for example

$$\Pi_{x_1x_1} = \frac{2}{3}\mu(2e_{x_1x_1} - e_{x_2x_2} - e_{x_3x_3}),$$

where $e_{x_ix_i}$ is the rate of strain component and the details are given in [5]. The centrifugal and Coriolis forces are in the term of H.

The numerical solution of the equations is obtained by use of the Beam-Warming delta form. The approximate factorization is applied to the form for finding a steady solution as follows:

$$\left(1+\Delta t\frac{1}{r}\frac{\partial A_3}{\partial \theta}\right)\left(1+\Delta t\frac{\partial A_2}{\partial r}\right)\left(1+\Delta t\frac{\partial A_1}{\partial x}\right)\Delta U = -\Delta t\ L(U^{(n)}),$$

where $\Delta U = U^{(n+1)} - U^{(n)},\ A_i = \partial E_i/\partial U,\ (i = 1,2,3)$, that is

$$A_1 = \begin{pmatrix} 0 & 1 & 0 & 0 \\ -w_x^2 + RT & 2w_x & 0 & 0 \\ -w_x w_r & w_r & w_x & 0 \\ -w_x w_\theta & w_\theta & 0 & w_x \end{pmatrix},$$

$$A_2 = \begin{pmatrix} 0 & 0 & 1 & 0 \\ -w_x w_r & w_r & w_x & 0 \\ -w_r^2 + RT & 0 & 2w_r & 0 \\ -w_r w_\theta & 0 & w_\theta & w_r \end{pmatrix},$$

$$A_3 = \begin{pmatrix} 0 & 0 & 0 & 1 \\ -w_x w_\theta & w_\theta & 0 & w_x \\ -w_r w_\theta & 0 & w_\theta & w_r \\ -w_\theta^2 + RT & 0 & 0 & 2w_\theta \end{pmatrix}.$$

The practical method of solution is not for θ but for the variable $\theta' = \theta - \theta_0(x), \theta_0(x) = (L-x)/R_2 \tan\alpha$, where R_2 is the outer radius of the rotor and α the inclination angle of the groove (Fig. 2). Then the circumferential calculation is independent of x. The boundary conditions are non-slip ones on the rotor and the casing. But a periodic boundary condition is applied for the gap ends, since only one groove of the rotor is simulated.

RESULTS AND DISCUSSION

Numerical calculation is performed for one of the spiral grooves. The geometry is as follows (see Figs. 2 and 3) :

outer radius of a rotor $R_2 = 68.8$ mm, inner radius $R_1 = 64.8$ mm and groove height $h = 4$ mm,
groove width $a' = 50.5$mm ($\sim 13.06/\sin\alpha$) , 1 pitch on the radius $R_2 = 74.14$ mm, $\alpha = 15°$,
gap : $\delta = 0.6$mm,
axial length $L = 115$ mm.

Gas is nitrogen with the viscosity coefficient of $\mu = 1.786 \times 10^{-5}$Pa·s of the constant temperature 300K. The number of revolutions per minute n is 18000 rpm. The exit pressure is fixed with the value of $p = P_2 = 1$ Torr. The inlet pressure is $P_1 = 1,\ 0.75$ and 0.5 Torr.

The number of the grid points in which the reasonable results are obtained is $101 \times 70 \times 61$ in x, r and θ directions respectively. The points for the groove region are $101 \times 61 \times 43$ and those for the gap $101 \times 9 \times 61$. The time step Δt is determined so that the CFL condition is fully satisfied. In this problem, the flow is under an adverse pressure gradient. If the CFL number is not very small in comparison with 1, the solution is immediately broken at some time step. Then we choose $\Delta t = 1.0 \times 10^{-7}$ sec.

The flow rate Q in SCCM unit (standard cubic centimeter per minute : atm-cm^3/min) and is expressed by the equation

$$Q = -\frac{6 \times 10^6}{\rho_0}\int\!\!\int_S \rho w_x r\ dr d\theta,$$

where S is the cross sectional area normal to the x-axis, ρ_0 the density of nitrogen at the standard condition and all quantities except Q measured in MKS unit. The mean pressure $\bar{p}$ over the cross section is also given by the similar equation as follows:

$$\bar{p} = \frac{1}{S}\int\!\!\int_S pr\ dr d\theta.$$

Calculation was carried out for one of the spiral grooves and the pumping performance of the turbo-molecular pump was estimated for six grooves.

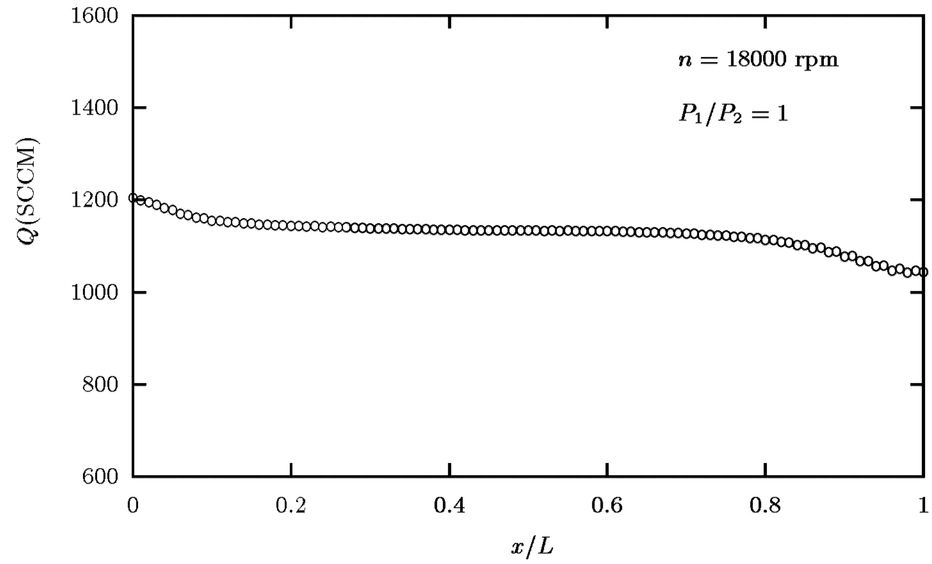

FIGURE 4. Averaged flow rate distribution for $P_1/P_2 = 1$

Figure 4 shows the local flow rate distribution averaged over the cross section for $P_1 = P_2 = 1$ Torr at $n = 18000$ rpm. Here and what follows, x in the figure means $L - x$ in the basic equations, that is, the inlet and the exit are $x = 0$ and L , respectively. It is seen that the flow rate is nearly constant except the inlet and the exit region. Then we can consider the flow to be steady (see [4]). The detailed inspection shows that the distribution is oscillating. It is to be noted that a large factor (6×10^6) is multiplied to obtain the flow rate Q in SCCM. From the figure, the flow rate Q is about 1138 SCCM in the middle region. This is smaller than that of no-gap flow (1239.7 SCCM) [4]. The area of the calculation region with gap is about 21 % greater than that without gap. But the flow rate decreases for the configuration with a gap.

Figure 5 shows the pressure distribution $\bar{p}$ for $P_1 = P_2 = 1$ Torr at $n = 18000$ rpm. The pressure ratio is uniform for the whole region, but a little smaller than 1.

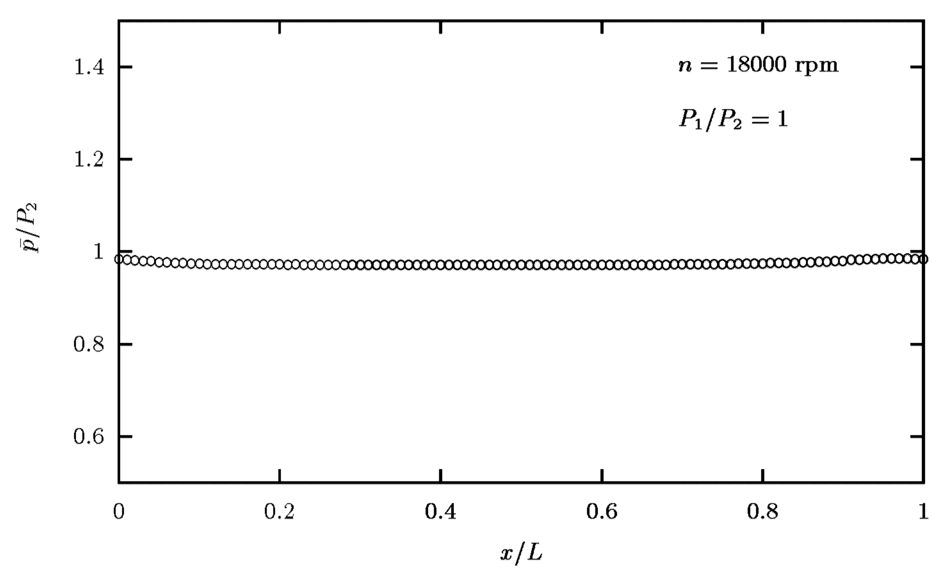

FIGURE 5. Averaged pressure distribution for $P_1/P_2 = 1$

Next we show the results of an adverse gradient flow. Figures 6 and 7 show the results of $P_1 = 0.75$. The flow rate distribution is given in Fig.6 and the pressure distribution in Fig.7. From Fig.6, it is seen that the

distribution is not uniform: the initial decrease followed by the gradual increase in the middle region and the decrease at the exit. The flow may not be in a steady state. But we may imagine the result of the final state from the figure. The pressure distribution, on the contrary, shows a gradual increase from 0.75 to 1, as seen from Fig. 7. Except the inlet and the exit region, a linear increase is seen. But the no-increase region at the inlet is very small. In fact, for the no-gap flow [4], the region extends to 1/3 of the whole groove and the sudden pressure rise appears for the region $x/L > 0.5$. Then the distribution was not linear. As was pointed by Sawada [7], the initial flat pressure distribution contributes to the flow rate increase and the pressure rise for the downstream region results in pressure difference.

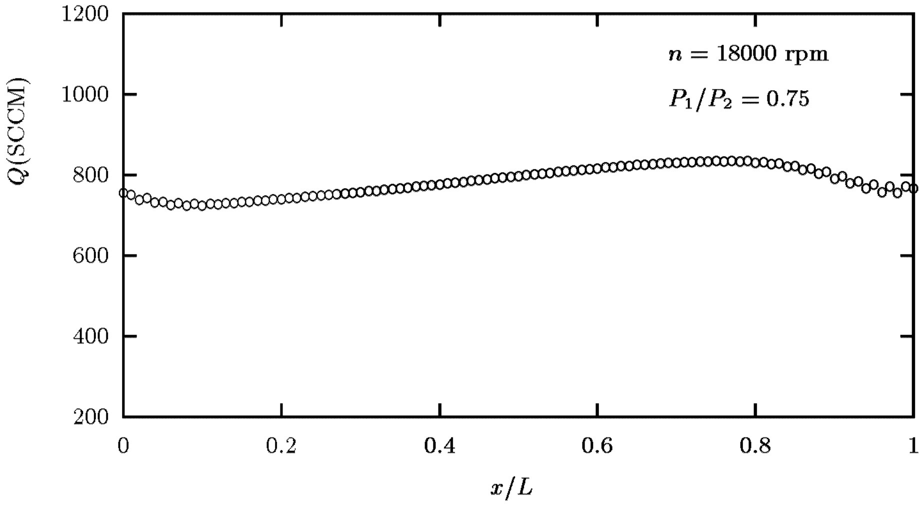

FIGURE 6. Averaged flow rate distribution for $P_1/P_2 = 0.75$

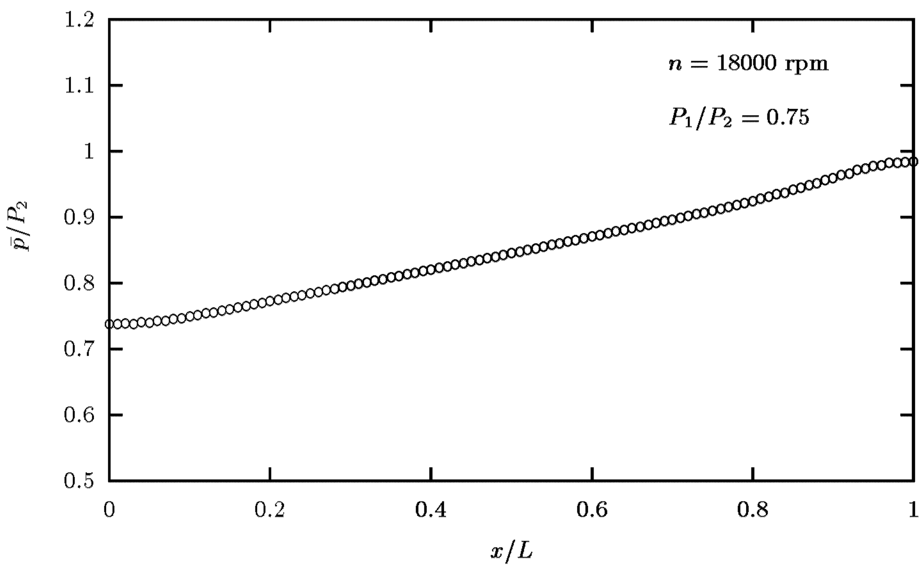

FIGURE 7. Averaged pressure distribution for $P_1/P_2 = 0.75$

Figures 8 and 9 show the results for $P_1 = 0.5$ Torr. The pressure distribution is given in Fig. 8. A linear increase in pressure ratio from 0.5 to 1 can be seen from the figure. The flow rate (Fig.9) is wavy and not uniform. This distribution has a similar one as that for $P_1/P_2 = 0.75$. If we further continue the simulation, we may consider that the maximum rate at the region of $x/L = 0.8$ decreases and that at the inlet and the exit region increases.

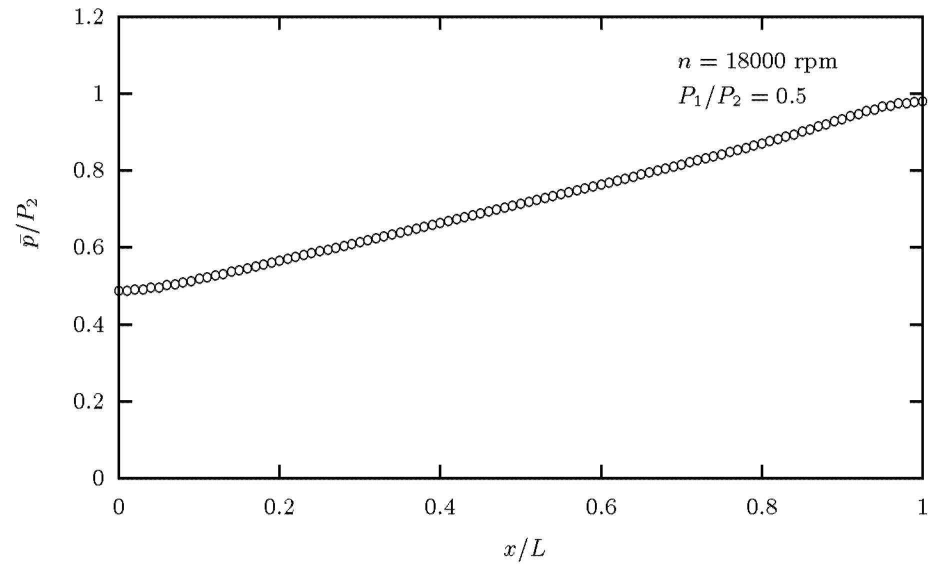

FIGURE 8. Averaged pressure distribution for $P_1/P_2 = 0.5$

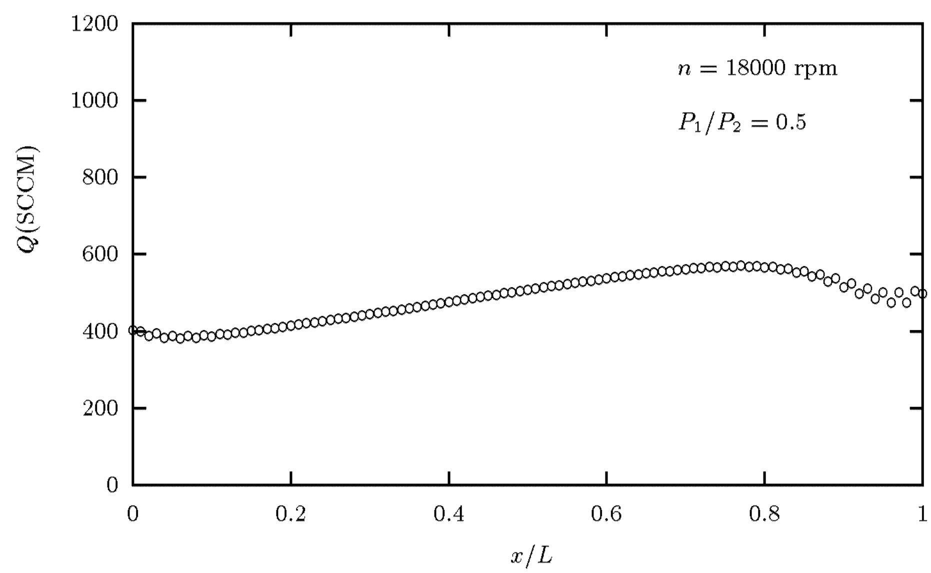

FIGURE 9. Averaged flow rate distribution for $P_1/P_2 = 0.5$

Figure 10 shows the pumping performance, that is, the pressure difference versus flow rate for $n = 18000$ rpm in comparison with the results of the no-gap flow [4], the channel flow approximation [2] and the experiment in [2]. The channel flow approximation has a tendency that the flow rate is overestimated especially for larger pressure difference. The no-gap result shows a larger flow rate for larger pressure difference. The present result is in a good agreement with the experiment for large pressure difference. If we don't take account of the gap effect, we estimate a flow rate with 10% larger value for $P_2 - P_1 = 0$ Torr and 20% one for $P_2 - P_1 = 0.5$ Torr.

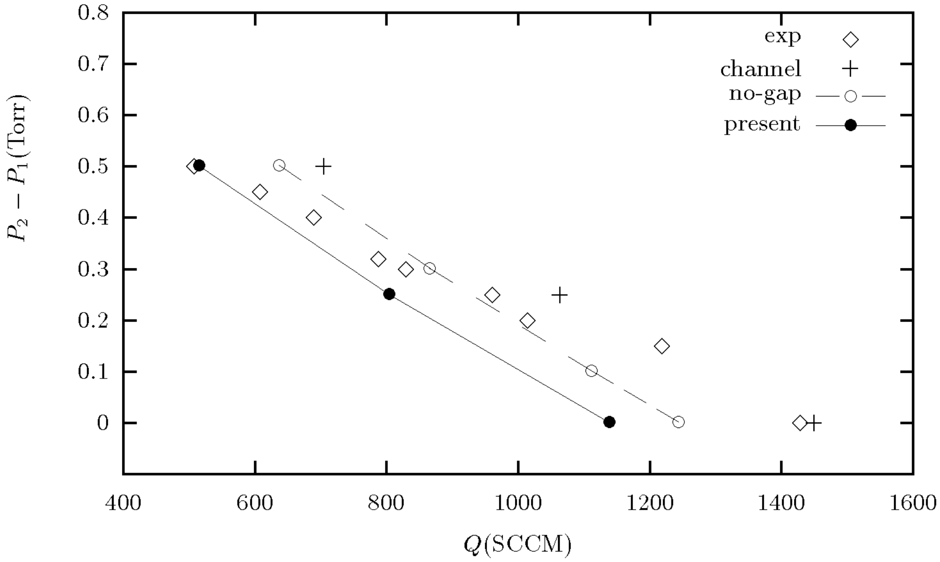

FIGURE 10. Pressure difference versus flow rate for $n = 18000$ rpm

We conclude that the clearance effect is significant for larger pressure difference. In this calculation, the number of revolution per minute n is 18000 rpm. As the speed increases, the effect of temperature change as well as the clearance is also important and must be investigated.

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